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PROGRESS REPORT

CONCEPTUAL DESIGN AND ANALYSIS OF A NOVEL ACTUATOR
by

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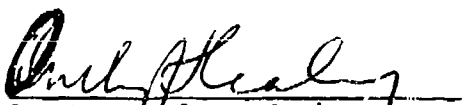
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NOMENCLATURE:

A area; tabulated constant
 β bulk modulus
Cr compression ratio
 c_v flow coefficient
d diameter
 ΔP pressure difference
F force
G specific gravity
J mass moment of inertia
 l length
m mass
P pressure
Q volumetric flow rate
R gas constant
 ρ density
t time
T temperature
 θ rotational coordinate
V volume
W weight
X linear coordinate

Subscripts

a atmosphere
H hydraulic

i inlet
o outlet
oh hydraulic orifice
P pneumatic
p piston
ph hydraulic piston
pp pneumatic piston
r rod
rh hydraulic rod
rp pneumatic rod
0 initial
1 coordinate 1; air supply
2 cylinder chamber 2; coordinate 2
3 cylinder chamber 3
4 cylinder chamber 4
5 cylinder chamber 5
11 linkage dimension
12 linkage dimension
13 linkage dimension

INTRODUCTION:

Robotic applications have increased steadily with advancing technology. Installation of manipulators to automate assembly line tasks has become commonplace because of the increased productivity, reliability, and cost savings which can be realized through their use. Projections estimate the value of the industrial robot population worldwide to be 10 billion dollars by the year 1990 [1]. The military is also interested in the application of robotics to missions where they may decrease risks to personnel and significantly enhance the probability of mission completion [2]. Further, cost effectiveness is a governing consideration in any use, paralleling the performance priorities of industrial applications.

A divergence in requirements between commercial and military robotic devices occurs because of the military need for mobility. While industrial manipulators are most often used in stationary installations such as factories, the nature of military missions dictate that a manipulator must be able to go to the location where its task will be accomplished, instead of having the job brought into its operating envelope. The mobility issue brings with it additional requirements for manipulator subsystems which are usually not considered in a stationary design. Some of these are power supply weight, system endurance, and ruggedness in changing environments.

Manipulator actuators must generally satisfy a larger number of functional criteria than similar power delivery devices used

in other systems. In addition to efficiency and reliability, the desirable qualities of the actuators for a robot arm may include high torque or force output throughout translation, quick response to signal orders, smooth reversibility, high stiffness with low power consumption when idle, positioning accuracy, and any other characteristics which are dictated by the functions which drive the total system design. Traditional choices for actuators have been confined mainly to electric motors and either hydraulic or pneumatic motors and cylinders. While each actuator category has been used successfully in manipulator applications, satisfactory conformity to the list of qualities needed is often difficult to achieve.

An innovative actuator type which may conform better to manipulator system requirements, particularly in mobile applications, is the stiff pneumatic-hydraulic actuator [3]. Figure 1 is a notional diagram of the construction of such an actuator in cross section, suitable for use in a low power, light load application [4]. The main motive power for this actuator would be provided by a low pressure compressed air source. In a mobile arm, this could conceivably be a rechargeable high pressure air tank with a regulator. The closed loop hydraulic side is a computer controlled self-contained circuit which adds motion dampening for precise position control during translation--a feature that pneumatic cylinders do not normally have. When ordered position is reached, the hydraulics add high stiffness, again an unusual feature for a low pressure pneumatic actuator.

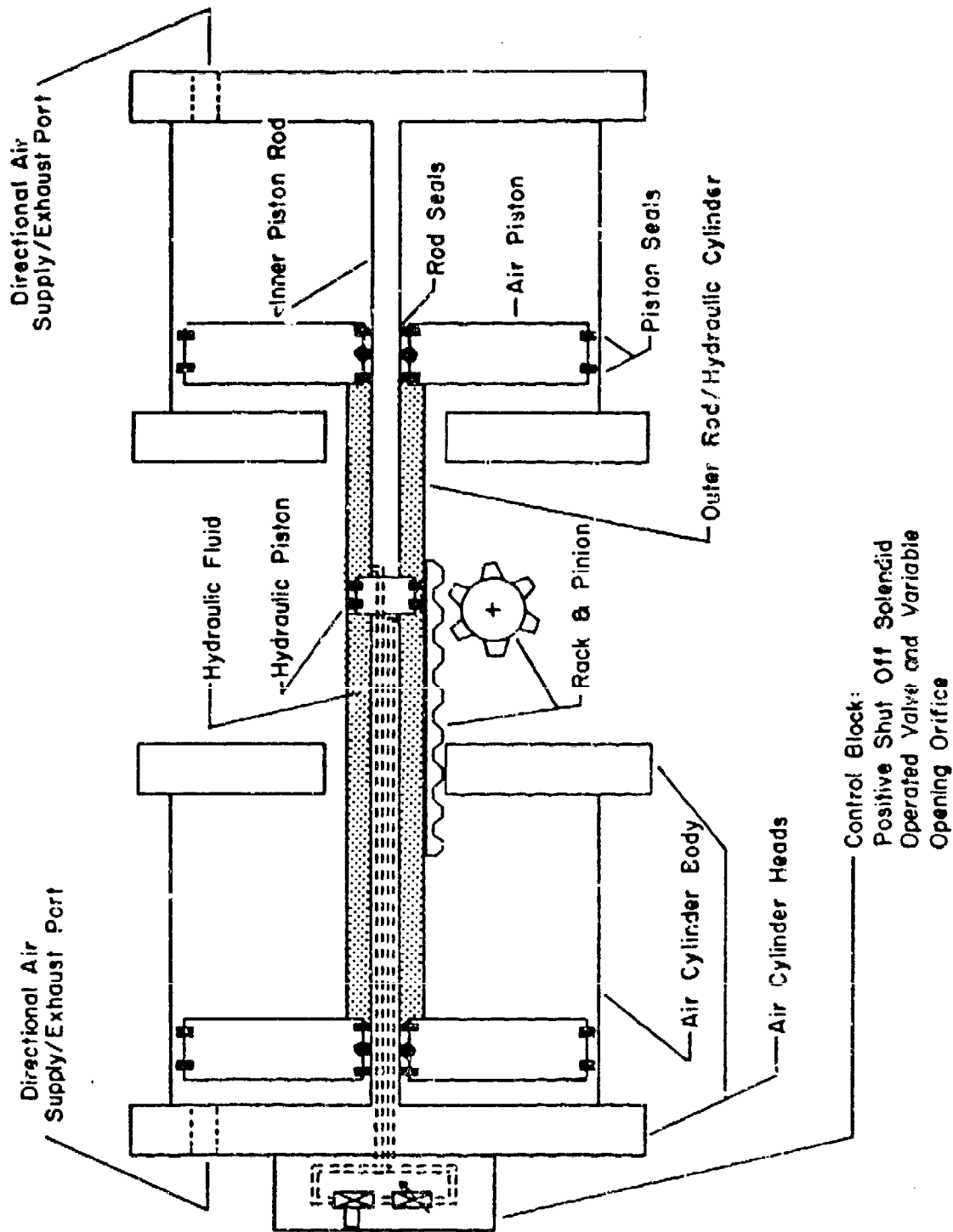


Figure 1 - Conceptual Novel Actuator

To investigate the suitability of this actuator for use on a mobile operational manipulator, a concept feasibility was conducted using the systems analysis approach and the functional-based design principles detailed by Blanchard and Fabrycky [5]. Following the concept analysis, a first order mechanical system analysis was developed from which system simulations will later be performed for the purpose of gathering data for prototype motion predictions and control design.

CONCEPT FEASIBILITY:

A system analysis of the stiff pneumatic-hydraulic actuator concept, hereafter known as the novel actuator, began by envisioning the actuator as a subsystem of a mobile operational manipulator, and developing a set of operational requirements for the system as a whole. The definition of system operational requirements was important because of all the design decisions which will be driven by these items. Without information on how and where the system will be used and what the system should be able to do, incorrect assumptions may be made and the resulting product may not accomplish its intended functions. After the requirements were determined, the relative merits of several alternative actuator subsystem choices were assessed to exhibit the potential of the novel actuator to fulfill the requirements. A broad view of the focus of the system design follows, with appropriate definitions.

The field of interest is actuation for a mobile operational manipulator. **Manipulator** means the system which accomplishes a dexterous task through its mechanism and the interaction of its different subsystems. Further, the task it accomplishes is one that can or could be done by a human arm if it were strong enough, accurate enough, timely enough, and able to withstand the environment in which the task must be performed. **Actuation** means the subsystem which applies forces or torques to the appropriate joints of the manipulator. **Mobility** implies that supporting systems must be compatible with the requirement that there be

some degree of independence from any one location.

To determine the operational requirements of a mobile arm, the type of manipulation which it would perform needed to be fixed. Because the problem of interest is the feasibility of an actuation concept, the chosen motion was a simple "pick and place" operation rather than a more complex assembly task or path following movement. The largest body of data on robotic manipulators has been collected on industrial applications [1,6,7,8]. The basis for the following requirements came from review of a survey of current industrial robots which yielded plausible characteristics for a typical arm to be adapted to the mobile concept [1].

The proposed initial Operational Requirements for a mobile manipulator system were:

1. Mission. To move and place a 5 lb payload using an estimated 5 lb manipulator structure in support of the manipulator system performance requirements. Total load to move and place is 10 lb.

2. Performance Parameters. The arm will be able to support:

Load rating - 10 lb.

Accuracy - positioning within .20 inch.

Speed - joint motion of 1 radian per second.

Reliability - 600 to 700 hours mean time between failure.

In addition, the actuator will help provide stiffness to the manipulator, a desirable quality which should be inherent in the

actuation system. The actuator will be serviceable on the manipulator and replaceable on-site.

3. Deployment. The manipulator will be deployed on a mobile platform and will perform material handling tasks such as bin picking, positioning, and palleting.

4. Operational Life Cycle. It is estimated that within 6 years, the manipulators will be ready to be replaced because of wear and/or technology changes.

5. Utilization. There will be intermittent 24 hour per day usage of the manipulator in changing environments. Based on an arbitrary 600 hours continuous operation for 4 months, the continuous operating hours per day will be $600/120 = 5.00$ hours.

6. Effectiveness Factors. Mean time between maintenance (MTBM) will be 600 operating hours, or servicing will be done every four months.

Mean time between failure (MTBF) will be 700 operating hours. Failure will occur after $(700 \text{ hrs}/600 \text{ hrs/yr})(4 \text{ mos}) = 4.67 \text{ mos}$ if no maintenance is performed.

7. Environment. The system will operate in an environment of ambient air with the following worst case qualities:

- 14 - 140 F
- 90% relative humidity
- airborne particles (dust, dirt)
- possible contact with explosive gasses and liquids
- shock excursions due to platform motion

from acceleration of sea state up to
state six

A general specification chosen for a manipulator conforming to the operational requirements is:

1. General system description. A six degree of freedom pick and place manipulator with a four foot operating envelope capable of being operated from a mobile platform.
2. Operational Requirements. Able to move a total 10 lb load (payload plus manipulator mass). The mean time between failures will be 700 operational hours, and the total lifespan will be six years.
3. Maintenance concept. To be determined.
4. System Functional Block Diagram and Interfaces. See Figure 2.
5. Performance characteristics. In support of the rated payload of 5 lb, the manipulator will be able to meet accuracy in placement of ± 0.20 inch in three positional degrees of freedom. Also, the point to point placement time within the manipulator envelope will not exceed 3 seconds. The actuators will not detract from the arm stiffness when a holding position is reached.
6. Actuator physical characteristics. To support a medium sized manipulator with a four foot operating envelope, the actuator size will be capable of being confined within a cylindrical volume of four inch diameter and ten inch length.
7. Effectiveness specifications. The total manipulator cost

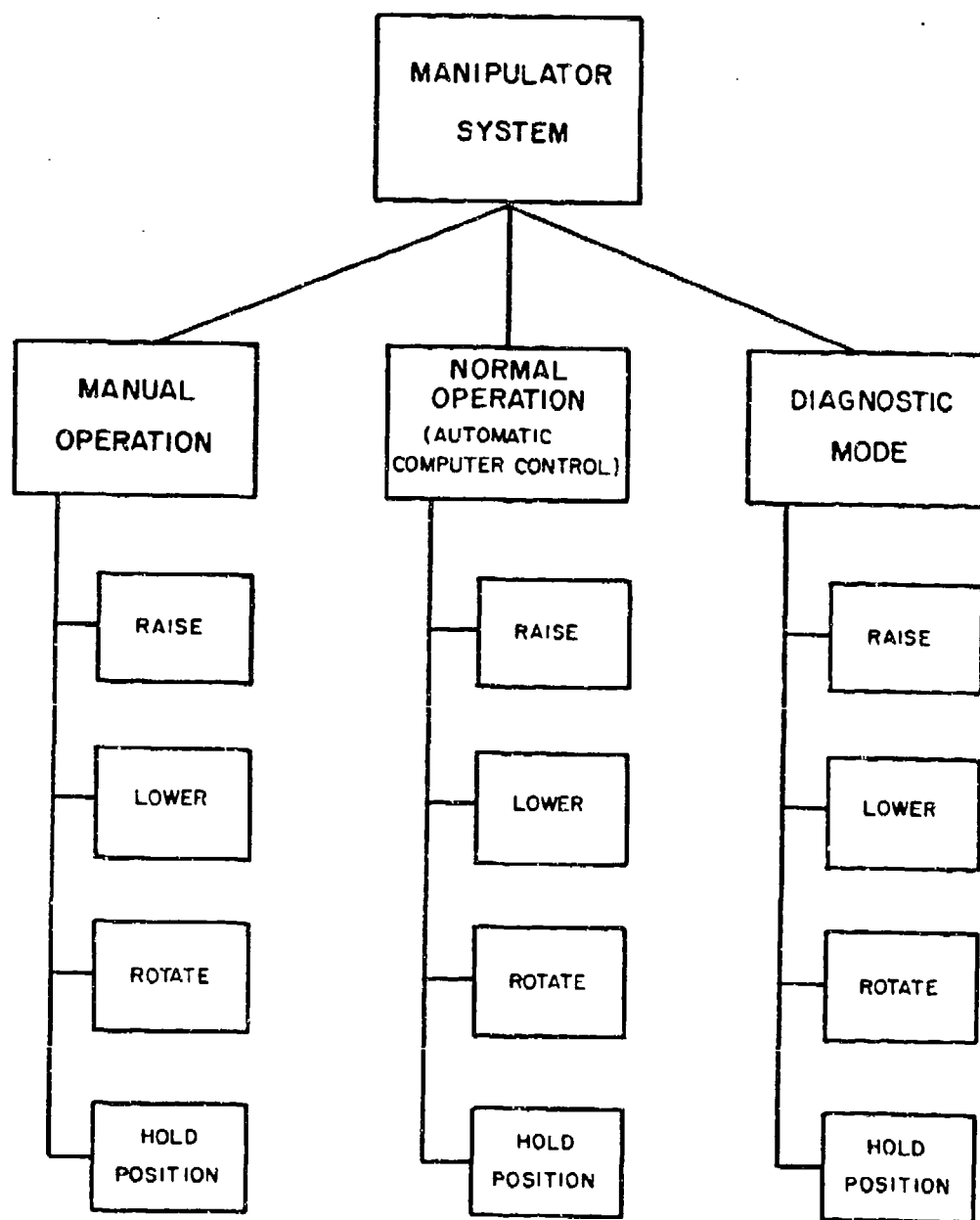


Figure 2 - System Functional Diagram

will be less than \$20,000.00. An estimate of outlay for manipulator actuators shall be \$5000.00. MTBF shall be 700 operational hours and replacement will be possible on site. The requirement for deployment on a moving platform means that the subsystems needed to accomplish the functions detailed on the functional relationship diagram must be self contained or "onboard". This may be the overriding consideration when hardware is chosen to satisfy the requirements.

The feasible actuation alternatives for a mobile operational manipulator were hydraulic, pneumatic, electric, or a novel combination. Each type is briefly discussed in the following sections.

Hydraulic actuators. They typically provide large force capability and higher power-to-weight ratios in fixed installations. They are used in high strength, stiff, precision control tasks such as drilling and other machine tool operations. The power plant required to operate the actuator would most likely be an electro-hydraulic configuration and as such would probably be heavier than other types of power plant. Other additional characteristics of hydraulic actuators are:

- a) Are able to withstand dirty, abrasive or wet operating environments and can tolerate temperature extremes well.
- b) Safe in explosive environments.
- c) Have a higher speed of response in general than electric motors.

- d) Usually small drop in speed with application of load.
- e) Cost of systems are high due to precision parts.
- f) Systems are susceptible to contamination.
- g) Systems are inherently messy because of unavoidable operating fluid leakage.
- h) Fire hazard of some fluid types is possible.
- i) They are heavy due to source equipment and fluid weight.

Pneumatic actuators. They are primarily found in simple manipulators. Typically, they provide uncontrolled motion between mechanical stops. They provide good performance in point-to-point motion, where they are simple to control and are low in cost. They would prove useful in applications such as end effector gripper devices and assembly line part rejection mechanisms. Device characteristics are:

- a) Simple and rugged, good in dirty or wet environments.
- b) Power is readily available in stationary applications; a portable reservoir is a viable choice for mobile platforms because it would be relatively lightweight.
- c) Safe in explosive environments.
- d) Compressibility of the actuating fluid detracts from stiffness, accuracy of positioning in mid-stroke, and response.

Electric actuators. They are relatively low in cost and they

interface easily to electronic drive circuits. They are used in low strength, precision applications such as manufacture of electronic circuit boards. Their characteristics are:

a) Electrical actuators are easy to activate and control.

b) They are compatible with electrical signal communication and the electrical power.

c) Good torque, speed, and continuous power output performance.

d) Relatively easy to use in servo-control applications because reversal of the rotor current reverses the motor torque and the motor behavior is good near zero torque or velocity.

e) Quiet and efficient in operation.

f) Unless they are direct drive at manipulator joints, they must operate at high speed through long, backlash-prone gear trains to generate powerful forces.

g) Power for mobile applications most likely comes from batteries which can be very heavy.

h) Some sort of brake is required to hold position if power use is not to be excessive.

i) The electric actuator is not as rugged as hydraulic and pneumatic actuators and cannot operate in dirty, abrasive, wet, and mildly corrosive environments unless they are sealed. Novel actuator. As envisioned, it would be used in a low power, moderate precision application. It would incorporate many

advantages of both pneumatic and hydraulic actuators with minimal drawbacks. Its characteristics would be:

- a) Excellent tolerance to dirty environments.
- b) Low system cost because of minimal high precision part manufacture; well within existing manufacturing capabilities.
- c) Excellent stiffness and positioning accuracy because of hydraulic components.
- d) Minimal contact with hydraulic fluid since it is in a self-contained system.
- e) Good candidate for mobile applications from a weight standpoint.

Clearly, the novel actuator optimized the advantageous characteristics of two separate actuator types while negating their respective disadvantages. This validated the selection of the novel actuator as the choice for an operational mobile manipulator.

MECHANICAL SYSTEM ANALYSIS:

A prototype to demonstrate concept operation for patent purposes was built as shown in Figures 3 and 4. However, in order to develop the original apparatus for research, several modifications were made. These included manipulator structural modifications, addition of an optical encoder at the arm joint to provide position and velocity data, and replacement of the manually operated valving with valves capable of actuation by computer TTL logic signals (Figures 5 and 6). The operating air pressure was designed to be 90 psig. The major components of the research prototype were:

Air Cylinder - Clippard Minimatic UDR-173. 1 1/16 inch bore, 3 inch stroke, 5/16 inch rod diameter.

Hydraulic Cylinder - Clippard Minimatic H9D-3D. 9/16 inch bore, 3 inch stroke, 1/4 inch rod diameter.

Air Valve - Koganei, Ltd. 110 series 5 port 3 position closed center solenoid operated. Rated 14 scfm at 100 psig supply pressure.

Hydraulic Valve - Skinner solenoid operated. Rated 100 psig.

Optical Encoder - Sumtak model LBL, 2048 pulses per revolution.

In order to analyze the system, a simplified model of the prototype was constructed as shown in Figure 7. The two objectives of the physical system analysis were: first, to show that the actuating system was extremely stiff when the joint is stationary; second, to find the governing equation of the system

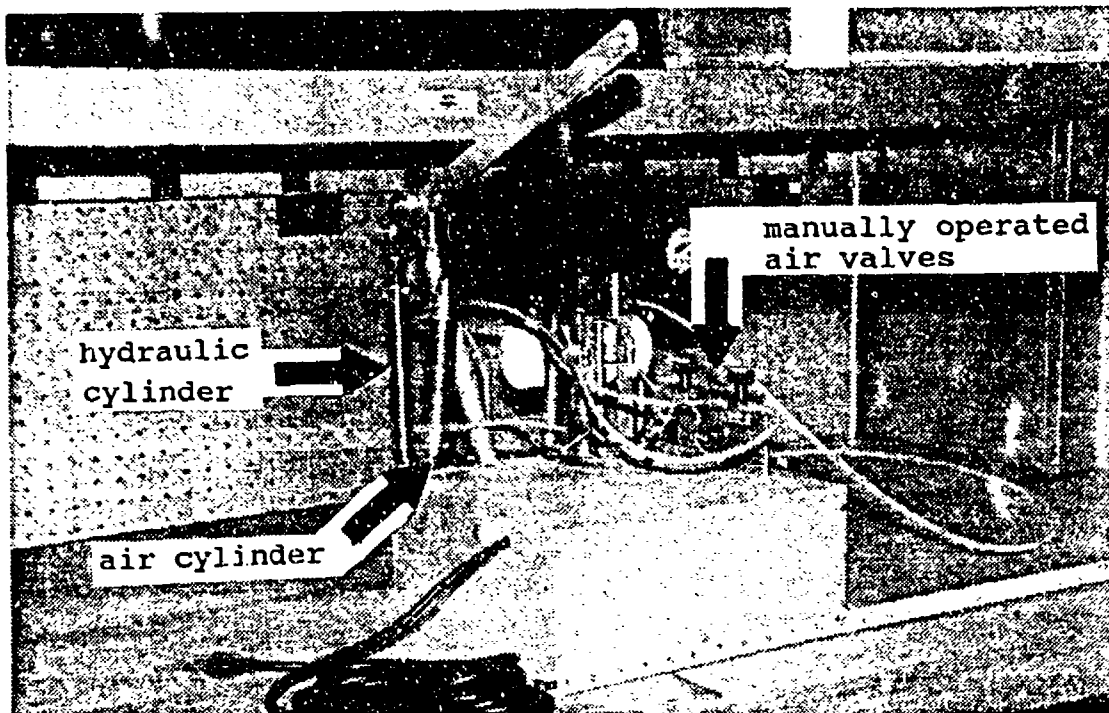


Figure 3 - Novel Actuator Prototype

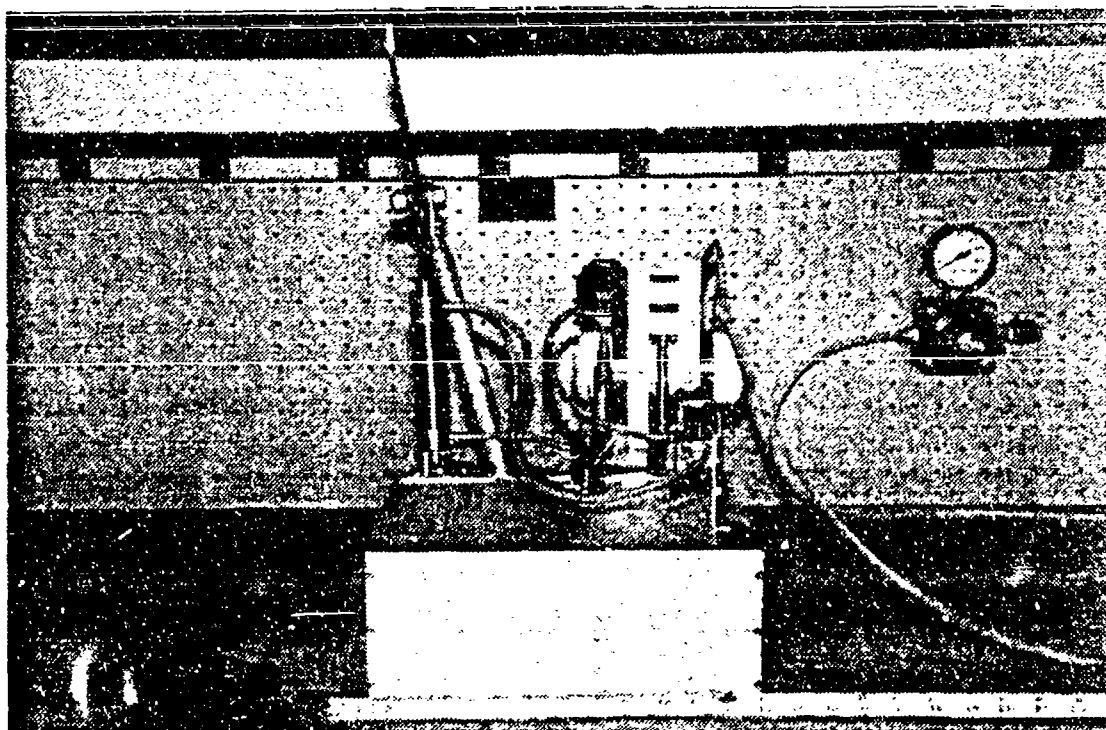
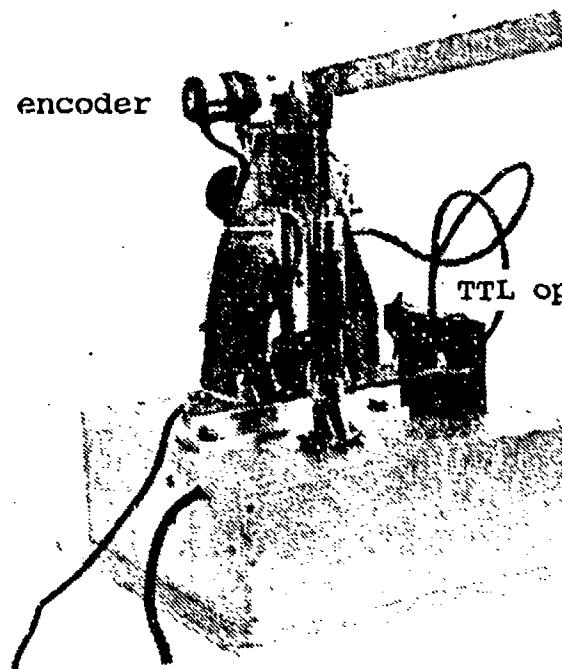


Figure 4 - Novel Actuator Prototype

position encoder



TTL operated air valve

Figure 5 - Modified Prototype

hydraulic solenoid valve
and needle valve

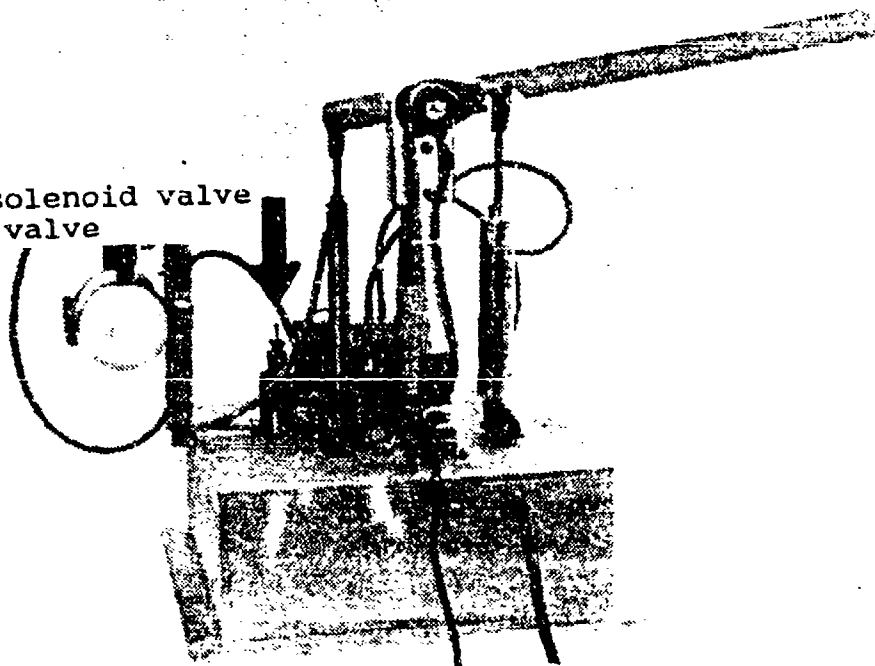


Figure 6 - Modified Prototype

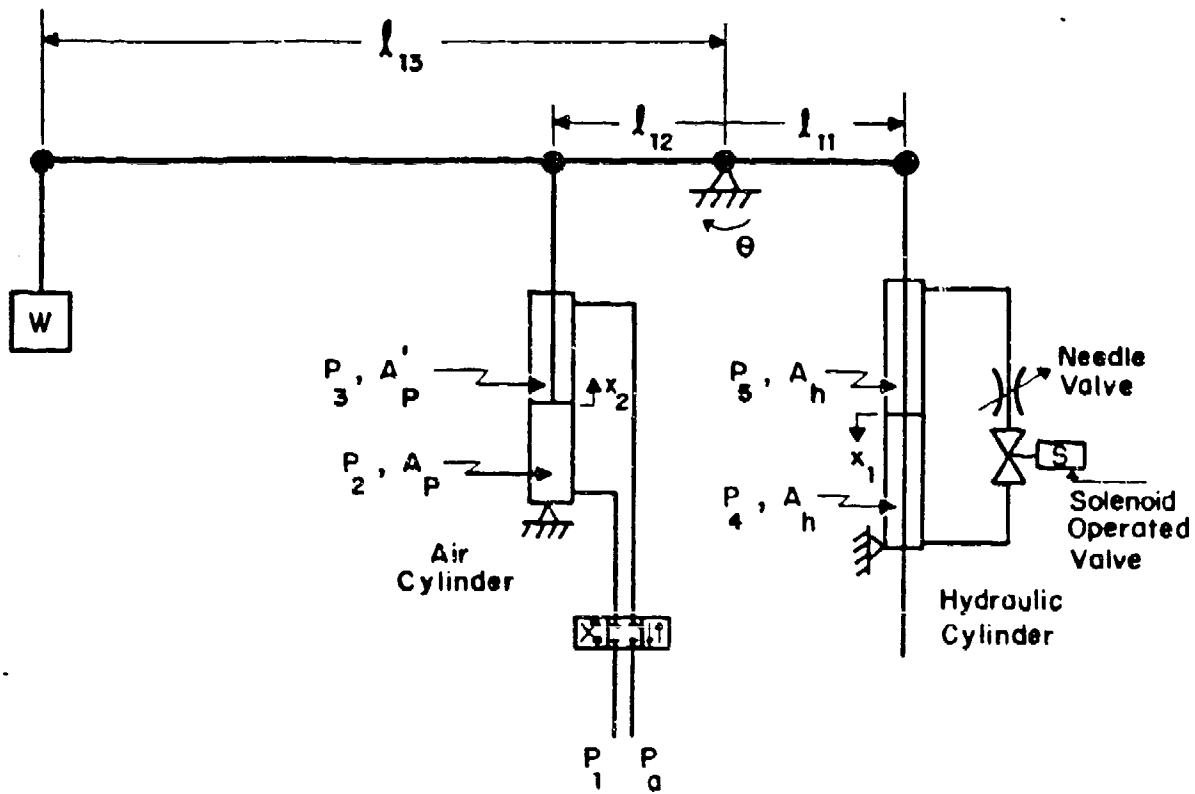


Figure 7 - Simplified System Model

dynamics so that simulations can be accomplished to predict motion and performance.

Stiffness Analysis. When the air cylinder was deenergized and the hydraulic solenoid valve closed, the system was motionless and the stiffness of the mechanism was defined in terms of the amount of deflection away from the no-load position when a load was applied to the arm. Consequently, assuming that the arm and linkage materials were sufficiently rigid so that structural deflection was negligible, the system stiffness was a function of the force of the load transmitted to the hydraulic cylinder and the bulk modulus of the hydraulic fluid. A summation of moments about the pivot point (Figure 7) gives:

$$\Delta P_H A_H \ell_{11} = W \ell_{13} \quad (1)$$

In turn, the pressure difference across the hydraulic cylinder can be expressed as:

$$\Delta P_H = \frac{W}{A_H} \frac{\ell_{13}}{\ell_{11}} \quad (2)$$

Since the rotary deflection of the mechanism can be expressed in terms of the linear deflection of the hydraulic piston, x_1 , further stiffness analysis focuses on the hydraulic cylinder as modeled in Figure 8. The assumption will be made that flow past the rod seals and piston seals is negligible. These leakage flows would be the only flows possible for this model, so:

$$Q = 0 \quad (3)$$

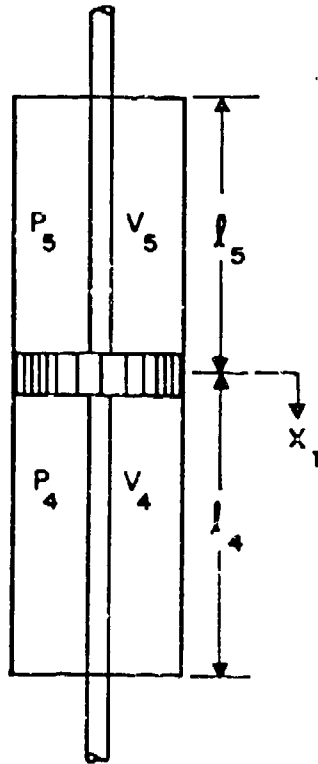


Figure 8 - Hydraulic Cylinder Model

The mass of fluid in each chamber of the cylinder is:

$$m = \rho V \quad (4)$$

Applying the continuity equation to the chambers:

$$\begin{aligned} \frac{dm}{dt} &= \frac{d(\rho V)}{dt} \\ \text{or} \quad \frac{dm}{dt} &= \rho \frac{dV}{dt} + V \frac{d\rho}{dt} \end{aligned} \quad (5)$$

The bulk modulus is defined as:

$$\beta \equiv \rho_0 \left(\frac{\partial P}{\partial \rho} \right)_T \quad (6)$$

Assuming that the pressure of the fluid is independent of temperature gives:

$$\beta = \rho_0 \frac{dP}{d\rho} \quad (7)$$

Substituting this into equation (5):

$$\frac{dm}{dt} = \rho \frac{dV}{dt} + V \frac{d\rho}{dt} \left(\rho \frac{dP}{d\rho} \frac{1}{\beta} \right)$$

or
$$\frac{dm}{dt} = \rho \frac{dV}{dt} + \frac{\rho V}{\beta} \frac{dP}{dt} \quad (8)$$

Dividing this resultant by ρ gives an equation for Q :

$$Q = \frac{dV}{dt} + \frac{V}{\beta} \frac{dP}{dt} \quad (9)$$

The assumption that $Q = 0$ yields:

$$\frac{dV}{dt} = - \frac{V}{\beta} \frac{dP}{dt} \quad (10)$$

For each chamber of the hydraulic cylinder:

$$\frac{dV_4}{dt} = - \frac{V_4}{\beta} \frac{dP_4}{dt} , \quad \frac{dV_5}{dt} = - \frac{V_5}{\beta} \frac{dP_5}{dt}$$

or
$$\frac{dV_4}{V_4} = - \frac{1}{\beta} dP_4 , \quad \frac{dV_5}{V_5} = - \frac{1}{\beta} dP_5 \quad (11)$$

Integrating both sides of each equation:

$$\ln\left(\frac{V_4}{V_{04}}\right) = - \frac{1}{\beta} (P_4 - P_{04}) , \quad \ln\left(\frac{V_5}{V_{05}}\right) = - \frac{1}{\beta} (P_5 - P_{05}) \quad (12)$$

Now, assuming

$$P_{04} = P_{05} = P_0 , \quad V_{04} = V_{05} = V_0$$

(thus $\ell_4 = \ell_5 = \ell$ as in Figure 8)

Solving each equation for P_0 then combining yields an expression

for the pressure difference across the piston:

$$\begin{aligned}
 P_0 &= P_4 + \ln\left(\frac{V_4}{V_0}\right) \beta, & P_0 &= P_5 + \ln\left(\frac{V_5}{V_0}\right) \beta \\
 P_4 + \ln\left(\frac{V_4}{V_0}\right) \beta &= P_5 + \ln\left(\frac{V_5}{V_0}\right) \beta \\
 P_4 - P_5 &= \beta \ln\left(\frac{V_5}{V_4}\right)
 \end{aligned} \tag{13}$$

$$V_5 = A(\ell + X_1), \quad V_4 = A(\ell - X_1) \tag{14}$$

$$\begin{aligned}
 P_4 - P_5 &= \Delta P_H = \beta \ln\left(\frac{A(\ell + X_1)}{A(\ell - X_1)}\right) \\
 \Delta P_H &= \beta \ln\left(\frac{1 + X_1/\ell}{1 - X_1/\ell}\right)
 \end{aligned} \tag{15}$$

Solving for linear deflection X_1 and substituting for ΔP_H using equation (2) gives

$$\begin{aligned}
 X_1 &= \left(\frac{\exp(\Delta P_H/\beta) - 1}{\exp(\Delta P_H/\beta) + 1} \right) \ell \\
 X_1 &= \left(\frac{\exp(W \ell_{13} / A_H \ell_{11} \beta) - 1}{\exp(W \ell_{13} / A_H \ell_{11} \beta) + 1} \right) \ell
 \end{aligned} \tag{16}$$

An effective bulk modulus for the system can be found using the method described by Merritt [9]. A small volume of trapped air within the hydraulic system can be seen to greatly reduce the bulk modulus and thus the stiffness of the mechanism. If there is no entrained air in the system and a representative value for the bulk modulus of 220,000 psi for petroleum based hydraulic fluids is used in equation (16), the stiffness of this mechanism can be seen to be excellent.

Dynamic Motion Analysis. To find the governing equation of

motion for the prototype mechanism, the forces acting on the linkage are represented in figure 9 [10]. The summation of moments about point o results in

$$-J\ddot{\theta} - Wl_{13} - F_H l_{11} + F_P l_{12} = 0$$

$$\text{or} \quad F_P l_{12} - F_H l_{11} - Wl_{13} = J\ddot{\theta} \quad (17)$$

Some assumptions which simplify this analysis are:

a) The hydraulic solenoid valve and needle valve combination in the hydraulic line connecting the cylinder ports will be treated as a circular orifice through which flow is turbulent. Then the equation for volumetric flow through the

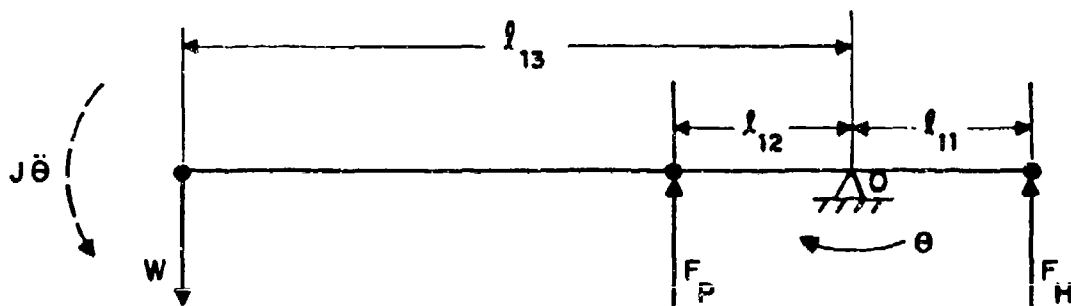


Figure 9 - Forces Acting on the Linkage

orifice which is developed by Merritt [9] will be applicable:

$$Q_H = 100 A_{oh} \sqrt{P_4 - P_5} \quad (18)$$

where Q_H = volumetric flow, in³/sec
 A_{oh} = orifice area in square inches
 P_4 = chamber 4 pressure, psig
 P_5 = chamber 5 pressure, psig

b) Flow through the air valve will be determined by the flow coefficient (c_v) method detailed in [11]. So Q for the air valve will be

$$Q_p = 22.48 c_v \sqrt{\frac{\Delta P (P_2 - P_a)}{T_1 G}} \quad (19)$$

where Q_p = volumetric flow, scfm
 c_v = flow coefficient
 $\Delta P = P_2 - P_1$
 P_1 = valve supply pressure, psig
 P_2 = valve outlet pressure, psig
 P_a = atmospheric pressure, psia
 T_1 = supply temperature, °R
 G = specific gravity of the flowing medium;
 1.0 for this analysis

c) Volume flows in the air cylinder, and through the air valve, are referred to standard cubic feet per minute (scfm). For a cylinder

$$Q = \frac{V Cr}{28.8 t} \quad (20)$$

where V = volume of the cylinder, cubic inches
 Cr = compression ratio = $(\text{inlet press} + 14.7)/14.7$
 t = time to fill cylinder

When one chamber of the air cylinder is energized, the flow out of the opposing chamber to atmosphere is dependent upon the compression ratio generated by the flow through the restriction of the valve and line to atmosphere. The pressure difference between the chamber and atmosphere is in turn dependent upon the piston speed and the size of the exhaust restriction. In an efficient application, a ΔP of 2 psi is acceptable and will be assumed, giving a Cr on the opposing chamber side of 1.14.

d) For this analysis, mechanism motion is taken to be in the positive θ , X_1 , and X_2 directions.

e) Air follows the perfect gas equation of state

$$Pv = RT \quad (21)$$

Figure 10 shows the forces acting on the air piston. The piston areas are:

$$A_p = \frac{\pi d^2}{4} \quad (22)$$

$$A_p' = A_p - A_{rp} \quad (23)$$

where $A_{rp} = \text{rod area} = \frac{\pi d_r^2}{4}$

So summing the forces yields

$$A_p P_2 - A_p' P_3 - F_p - m_{pp} \ddot{X}_2 = 0 \quad (24)$$

Assuming the mass of the piston is negligible

$$F_p = A_p P_2 - A_p' P_3 \quad (25)$$

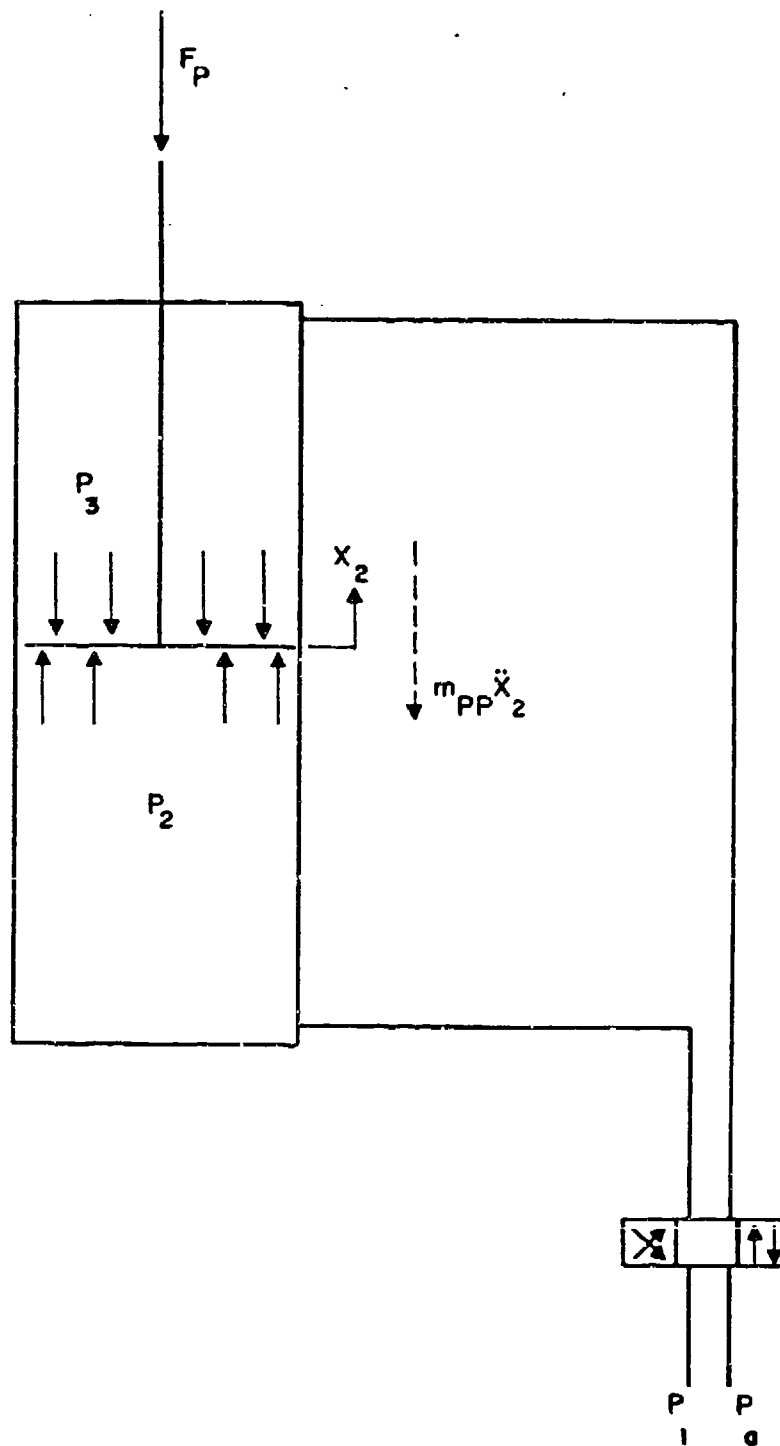


Figure 10 - Forces Acting on the Air Piston

The hydraulic piston is similarly depicted in Figure 11.

$$A_H = \frac{\pi d^2}{4} - A_{rh} \quad (26)$$

Summing the forces as before

$$F_H + A_H P_5 - A_H P_4 - m_{ph} \ddot{X}_1 = 0 \quad (27)$$

Assuming negligible piston mass

$$F_H = A_H (P_4 - P_5) \quad (28)$$

A continuity analysis of the air cylinder yields:

Flow into chamber 2

$$Q_2 = A_p \dot{X}_2 Cr_2 \quad (29)$$

Flow out of chamber 3

$$Q_3 = A_p \dot{X}_2 Cr_3 \quad (30)$$

also

$$Q_3 = (A_p - A_r) \dot{X}_2 Cr_3 \quad (31)$$

so combining equations (29) and (30)

$$Q_2 = Q_3 (Cr_2/Cr_3) + A_r \dot{X}_2 Cr_2 \quad (32)$$

Flow through the valves is calculated using equation (19).

The coefficients of c_v have been tabulated in [11] for different supply pressures and pressure drops through the valve, and is called constant "A." So,

$$Q = c_v/A \quad (33)$$

Flow through the inlet portion of the valve

$$Q_i = c_{vi}/A_i \quad (34)$$

Flow through the outlet portion of the valve

$$Q_o = c_{vo}/A_o \quad (35)$$

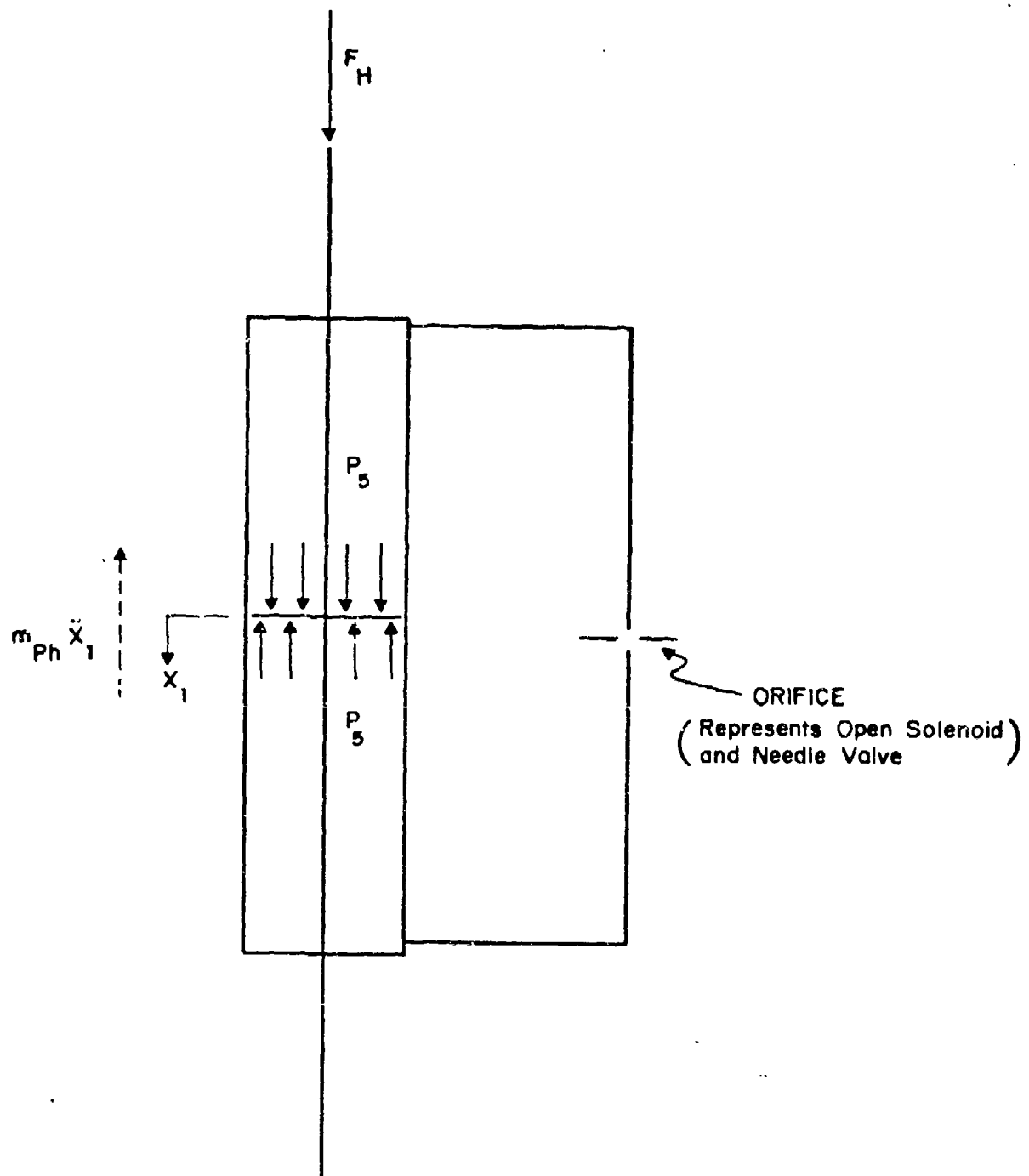


Figure 11 - Forces Acting on the Hydraulic Piston

Because of continuity

$$Q_1 = Q_2$$

and

$$Q_3 = Q_o \quad (36)$$

The preceding analysis for the air cylinder provides a means of sizing the air valve to meet a desired flow requirement based on an operating speed, supply pressure, and pressure drop across the valve. Since the actuator control issues appear to revolve about the hydraulic portion of the system, it will be assumed that the air valve will be sized to meet or exceed the flow required for a maximum speed, which occurs when the hydraulic orifice A_{oh} is open to its maximum area so that ΔP is a minimum. Continuity analysis for the oil cylinder:

Flow into chamber 5 equals flow out of chamber 4, so

$$Q_5 = Q_4 = A_H \dot{X}_1 \quad (37)$$

Flow through the orifice is calculated using equation (18). To maintain dimensional consistency, convert Q_H to cfm

$$Q_H = \frac{100}{28.8} A_{oh} \sqrt{P_4 - P_5} \quad (38)$$

Because of continuity

$$Q_H = Q_4 = Q_5 \quad (39)$$

In the above equation, Q must also be converted to cfm

$$Q_5 = Q_4 = \frac{A_H \dot{X}_1}{28.8} \quad (40)$$

The hydraulic cylinder analysis provides a way to determine the pressure drop across the piston for a given operating speed

and orifice opening.

Figure 12 shows the geometric compatibility of the mechanical linkage. From this figure, the following relations are derived:

$$\theta = x_1/l_{11} = x_2/l_{12} \quad (41)$$

so

$$\dot{\theta} = \dot{x}_1/l_{11} = \dot{x}_2/l_{12} \quad (42)$$

and

$$\ddot{\theta} = \ddot{x}_1/l_{11} = \ddot{x}_2/l_{12} \quad (43)$$

also

$$x_2 = \frac{l_{12}}{l_{11}} x_1$$

and

$$\dot{x}_2 = \frac{l_{12}}{l_{11}} \dot{x}_1 \quad (44)$$

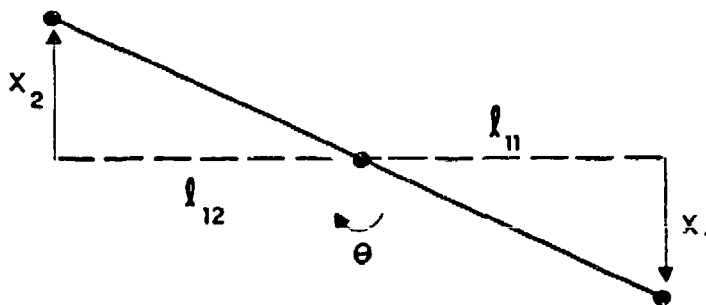


Figure 12 - Linkage Geometric Compatibility

Using the compatibility relations and the preceding force and continuity analyses, equation (17) can be recast into a form containing one variable so that numerical simulations can be performed to determine system dynamics. So

$$F_P \ell_{12} - F_H \ell_{11} - W \ell_{13} = J \ddot{\theta}$$

By geometric compatibility

$$F_P \ell_{12} - F_H \ell_{11} - W \ell_{13} = J \ddot{X}_2 / \ell_{12} \quad (45)$$

Substituting equations (25) and (28) into the above

$$(A_P P_2 - A_P' P_3) \ell_{12} - A_H (P_4 - P_5) \ell_{11} - W \ell_{13} = J \ddot{X}_2 / \ell_{12} \quad (46)$$

Applying geometric compatibility to equation (40) gives

$$Q_5 = Q_4 = \frac{A_H \dot{X}_2}{28.8} \frac{\ell_{11}}{\ell_{12}} \quad (47)$$

Then using equations (38) and (39)

$$\frac{A_H \dot{X}_2}{28.8} \frac{\ell_{11}}{\ell_{12}} = \frac{100}{28.8} A_{oh} \sqrt{P_4 - P_5}$$

or

$$A_H \dot{X}_2 \frac{\ell_{11}}{\ell_{12}} = 100 A_{oh} \sqrt{P_4 - P_5} \quad (48)$$

Solving for $P_4 - P_5$

$$P_4 - P_5 = \left(\frac{A_H \dot{X}_2 \ell_{11}}{100 A_{oh} \ell_{12}} \right)^2 \quad (49)$$

Finally, substitution of equation (49) into equation (46) yields

$$(A_P P_2 - A_P' P_3) \ell_{12} - A_H \left(\frac{A_H \dot{X}_2 \ell_{11}}{100 A_{oh} \ell_{12}} \right)^2 \ell_{11} - W \ell_{13} = J \ddot{X}_2 / \ell_{12} \quad (50)$$

Further simplification of the above equation can be achieved by assuming that the air valve is sized such that the portion of the first term representing F_p can be approximated by $A_p P_1$. This leaves the equation in the form

$$A_p P_1 \ell_{12} - A_H \left(\frac{A_H \dot{X}_2 \ell_{11}}{100 A_{oh} \ell_{12}} \right)^2 \ell_{11} - W \ell_{13} = J \ddot{X}_2 / \ell_{12} \quad (51)$$

This equation is nonlinear in X_2 . The use of an appropriate numerical simulation program will allow determination of mechanism position and velocity in response to given inputs of air supply pressure (P_1), applied load (W), and hydraulic modulating orifice opening (A_{oh}).

CONCLUSIONS AND RECOMMENDATIONS:

The novel actuator was determined to be a promising concept for application to robotic manipulators. Its capability to satisfy diverse and potentially opposing requirements for operation made it a more suitable choice than conventional actuators. Additionally, the advantages of relatively low weight, high stiffness, and ruggedness make it attractive for a mobile application.

A preliminary system analysis of an actuator prototype yielded an equation of motion with which dynamic simulations can be performed.

Recommended actions regarding further research on the novel actuator concept are:

1. Develop a computer simulation for prototype mechanism position and velocity prediction in response to typical inputs. Exercise the simulation to obtain data to compare with actual prototype response.
2. Refine the system analysis as necessary to find agreement between the simulation and the actual machine.
3. Develop a controller for the system so that point-to-point motion can be automatically controlled through the use of a computer.
4. Extend the design process to the detail design phase, from which a unit actuator and power supply capable of installation on a mobile manipulator could be produced.

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